MNK Publication

International Journal of Latest Research in Science and Technology **ISSN (Online):2278-5299**
Volume 1,Issue 1 : Page No.45 - 48 , May –June 2012
https://www.mnkpublication.com/journal/ijlrst/index.php

DESIGN OF CAPILLARY EXPANSION DEVICE USED IN VAPOR COMPRESSION REFRIGERATION SYSTEM

Prashant Sharma[1], Rahul Sharma[2]
[1]Associate Professor, Rajiv Gandhi University of Technology, Bhopal (M.P), India
[2]Assistant Professor, SIA, Indore (M.P), India

***Abstract* – Capillary tubes, which are used extensively in mechanical refrigeration system, have been standardized in terms of their parameters through analytical and experimental work. This work is mainly confined to straight capillaries. Since length required is usually large, other shape should also be considered for accommodation in available space. This work deals with analytical design of straight and with some modification helical shape of capillary devices.**

***Keyword* -** VCRS, Refrigeration, Capillary tubes

I. INTRODUCTION

Capillary tubes have found universal acceptance as expansion devices in refrigeration and air-conditioning units. They have no moving parts, unlike thermostatic expansion valves. With the availability of more versatile compressors, better refrigerants and nearly leak proof systems, the use of capillary tubes has become quite attractive and popular. The basic refrigeration and air-conditioning system basically comprises of two heat exchanging devices and two pressure variants.

The capillary expansion device provides the required pressure drop between the condenser and the evaporator and regulates the mass flow rate to meet the demand of the system. It is true that they cannot cope up with large variable load conditions and operate at maximum efficiency only at one set of operating conditions. At all other conditions, their efficiency is somewhat less than the maximum, but it should be borne in mind that they are self compensating devises to some extent and if properly designed and applied, will give satisfactory performance over a wide range of operating conditions.

Successful operation of capillary tubes depends on keeping the system fully charged with refrigerant and having an uniform internal diameter. Capillary tubes are basically small bore long tubes to provide required pressure drop. Internal cleanliness and dehydration should be maintained. Capillary tubes have been standardized in terms of diameter, length and thickness through much experimental and analytical work. Manufacturers there by recommend an inside diameter between 0.878mm and 1.4mm, a length between 1.5m and 4.9m and a thickness of the order of 0.635mm. The diameter has been maximized to reduce pressure equilisation time between the condenser and the evaporator during the "off cycle" period, permitting the use of a low-torque starting motor [1]. They also reduce sensitivity to dirt and frost deposits [2]. The recommended thickness prevents kinking. It has been reported that increased wall thickness increases the roughness [3]. Smooth tubes increase the mass flow rate by about 10% to 15% compared with rough tubes.

A numbers of graphs and curves are available in the literature for initial sizing of straight capillary tubes. Regarding other shapes, the only mention made is that a wound tube reduces the mass flow rate by about 5% due to increased resistance [4]. Also the curves available for straight capillaries are non-linear in nature and hence interpolation or extrapolation is to be carried out very carefully. Different investigators have proposed and also used different expression for frication factor in liquid and two phase zones. Property correlations used are also different. Mikol (1963) studied the flow of refrigerant in glass capillary tubes and suggested that in two phase zones equal velocities of liquid and vapour phase should be considered. This is called 'slug flow'. This aspect was confirmed by Koizumi and Yokohama (1980) in their experimental work. The frication factor expression proposed by Niaz and Davis is widely used, although they mentioned that the length they calculated using their frication factor gave 10% greater length than the experimental one for a given pressure drop. The other authors worked with their

Publication History

Manuscript Received	:	*15 April 2012*
Manuscript Accepted	:	*12 May 2012*
Revision Received	:	*27 June 2012*
Manuscript Published	:	*30 June 2012*

own friction factors so as to arrive at a length matching closely with the experimental results.

II. COMPUTATIONAL ANALYSIS

Figure 1 shows four basic elements of a vapour compression refrigeration system.

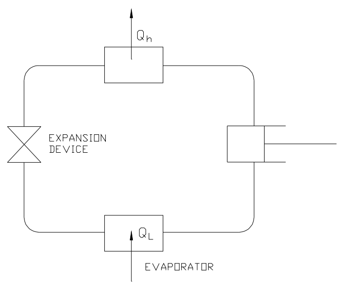


FIGURE 1 Basic vapour-comression refrigeration cycle

Quantities Qh and Ql are the amount of heat transferred in the condenser and evaporator respectively. Figure 2 shows four salient points (1-2-3-4) of a cycle on a pressure- enthalpy diagram.

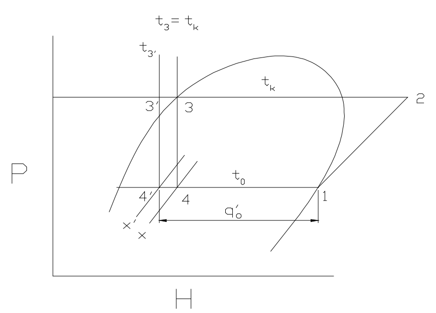


FIGURE 2 Effect of liquid subcooling

The process 3-4 stands for isenthalpic expansion of refrigerant with zero degree of sub cooling at the entry to the capillary tube while 3'-4' stands for entry with some degree of sub cooling. Normally in an ideal VCRS, the refrigerant entering the compressor must have some degree of superheat, while at entry to capillary expansion device, it must have a few degree of sub cooling as compressor can not handle two phase fluid while capillary can not effectively handle two phase entry. Otherwise both units will suffer due to mal-functioning.

The capillary tubes serve almost all small refrigeration systems and its applications extend up to a capacity of order of 10 kW. Numerous combination of bore and length are available to obtain the desired restriction. Once a capillary tube has been selected and installed, the tube can not adjust to variations in suction, discharge pressures or load. The compressor and expansion device must arrive at pressure conditions which allow the compressor to pump from the evaporator the same mass flow rate of refrigerant that the expansion device capillary feeds to the evaporator. A condition of unbalanced flow between these two components must necessarily be temporary. At high condensing pressures, the capillary tube feeds more refrigerant to the evaporator than it does at low condensing pressure because of the increase in pressure difference across the tube. The compressor and capillary tube do not have complete liberty to fix the suction pressure because the heat transfer relationships of the evaporator must also be satisfied. If the evaporator heat transfer is not satisfied at the compressor- capillary tube balance point, an unbalanced condition results that can starve the evaporator to overfeed the evaporator (fig.3)

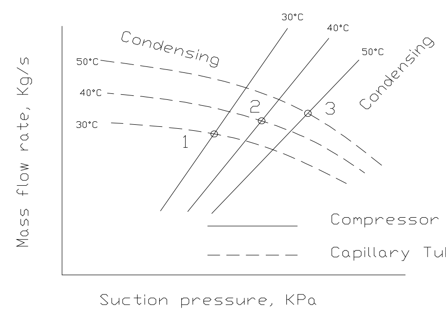
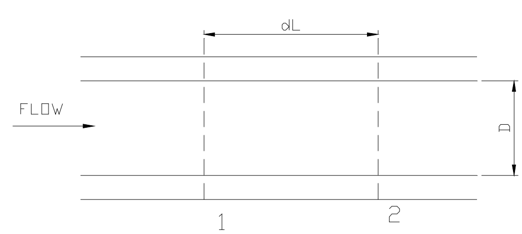


figure 3 Balance points with a reciprocating compressor and capillary tube.

II. ANALYTICAL COMPUTATION OF PRESSURE IN A CAPILLARY TUBE



The equation relating state and conditions at points 1and 2 in a very short length of capillary tube in the figure is written using following notions [5]

A: Cross sectional area of inside of tube, m^2

D: ID of tube, m.

f: friction factor, dimensionless

h: enthalpy, kJ/kg.

h_f : enthalpy of saturated liquid , kJ/kg

hg : enthalpy of saturated vapour, kJ/kg

ΔL: length of increment, m.

P: pressure, Pa

Re: Reynold's No., VD/υ

v : specific volume of m^3/kg

v_f : specific volume of saturated liquid, m^3/kg

v_g: specific volume of saturated vapour, m^3/kg

V: velocity of refrigerant, m/s

w : mass flow rate, kg/s

x : dryness friction

μ: Viscosity, $p_{a*}s$

μ_f: viscosity of liquid, $p_a \times_S$

μ_g: viscosity of Vapour, $p_a \times S$

The fundamental equations applicable to the control volume bounded by points 1and 2 in fig. are

1. Conservation of mass
2. Conservation of energy
3. Conservation of momentum

The equation of conservation of mass states that

$$w = V_1A/ v_1 = V_2A/ v_2 \ldots. \quad (1)$$

Or

$$w=V_1/ v_1 = V_2/ v_2 \ldots\ldots\ldots (2)$$

The conservation of energy gives

$$1000\ h1+ V^2{}_1/ 2 =1000\ h_{2+}V^2{}_2/ 2\ldots.. \quad (3)$$

This assumes negligible heat transfer in and out of system.

The momentum equation in words states that the difference in forces applied to the element because of drag and pressure difference on opposite ends of the element equals that is needed to accelerate the fluid.

$$[(p_1-p_2) - f\ \Delta L/D \quad V_2/ 2v]\ A = w\ (V_1-V_2) \ldots\ldots\ldots\ldots (4)$$

As the refrigerant flows through the tube, its pressure and saturation temperature progressively drop and the fraction of vapour 'x' continuously increases. At any point

$$h= h_f \ (1-x) + x\ h_g \quad \ldots\ldots\ldots\ldots (5)$$

And $\quad v= v_f(1-x) + x\ v_g \quad \ldots\ldots\ldots..(6)$

The quantities of eq^n (4) V, v and f all change as refrigerant flows from point 1to2. Simplifying using eq^n (2)

$$f\ \ \Delta L/D\ \ V_2/ 2v\ = \ f\ \ \Delta L/D\ V/ 2\ w/A \quad \ldots\ldots\ldots\ldots\ldots.(7)$$

In the calculation to follow, V used in eq^n (7) will be mean velocity

$$V_m = V_1+ V_2 / 2 \ldots\ldots\ldots.. (8)$$

The friction factor with turbulence is

$$F= 0.33/Re^{\ 0.25} = 0.33/(VD/\mu v)^{\ 0.25} \ \ldots\ldots(9)$$

The viscosity in two phase flow is given by

$$\mu = \mu_f\ (1-x) + x\ \mu_g \ldots\ldots.. (10)$$

The mean friction factor f_m applicable to incremental length 1-2 is $f_m = f_1+f_2/2 =[0.33/Re1^{\ 0.25} + 0.33/Re2^{0.25}]/ \ 2\ldots\ldots (11)$

IV EVALUATION OF INCREMENT LENGTH ΔL.

The essence of the analytical calculation is to determine the length ΔL between points 1-2 as shown in fig. for a given reduction in saturation temperature of the refrigerant. The flow rate and other conditions at point 1 are known and

For a required selected temperature at point 2, The remaining conditions at point 2 and ΔL would be computed in the following steps :

1. Temperature t_2 selected
2. p_2, h_{f2}, h_{g2}, v_{f2}, and v_{g2} are computed, all being function of temperature(or pressure).
3. combination of equation

   (2) and (3) gives

$$1000\ h_{\ 2+}\ v^2{}_2/ 2\ (w/A)^2 =1000\ \ h_1+$$

$V_1^2/ 2\ldots\ldots\ldots (12)$ Substituting equations (5) and(6) into (12)

$$1000\ h_{f2} +1000(h_{g2}- h_{f2})\ x +[\{ v_{f2}+(v_{g2} - v_{f2})x \}^2(w/A)^2] = 1000\ h_1 + V_1^2/ 2 \qquad \ldots\ldots\ldots\ldots\ldots.(13)$$

In equation, all quantities being knows except x, witch could be solved by quadratic equation,

$$X=[-b+\sqrt{b^2-4ac}]/2a\ldots\ldots..(14)$$

Where, $a= (v_{\ g2}- v_{\ f2}\ )^2\ (w/A)^2 \times 1/2$

$b=1000(h_{g2}- h_{f2})+ v_{\ f2}\ (\ v_{\ g2} - v_{\ f2})(w/A)$ and $c = 1000(h_{f2}-h_1)+(w/A)^2\ 1/2\ v_{\ f2}{}^2- V_1^2/2$

4. With the known value of x, h_2, v_2 and V_2 can be computed.
5. Reynolds No. is computed at point 2 using the viscosity from eq^n (10), the friction factor at point 2 from eq^n (9), and friction factor for increment length from eq^n (11)
6. Finally, substituting values from eq^{ns} (7) and (8) ΔL is evaluated.

V CONCLUSION

Analytical method being tedious, a computer program could be formulated to arrive at required length giving all the required details at the end of the capillary tube.

REFERENCES

[1] Bolstad, M.M. and R.C. Jordan. "Theory and use of the capillary tube expansion device" Refrigerating Engineering, Vol. 56, No. 12, pp.519-523

[2] Linge, K 1949. "The capillary tube as throttle device in refrigerators" Kaltetechnik (in German), Vol.1, No. 6, pp 125-129

[3] Sweedyk, J.M.1981. "Capillary tubes- Their standardization and use." ASHRAE Transaction, Vol.87, part 1, pp. 1062-76

[4] Kipp, E. and H.Schmidt. 1961 "Sizing of a capillary tube as a throttling device in refrigeration machinery" Kaltetechnik, Vol.13 No. 2, pp 46-51.

[5] Stoecker W.F., and Jones, J.W. "Refrigeration and Air conditioning" TATA McGraw-Hill pub. Co.Ltd.pp. 264-67.

[6] ASHRAE, 2002, Adiabatic capillary tube selection, Refrigeration Handbook, ch. 45, pp.45.26-45.30, ASHRAE.

[7] Bittle R.R., Oliver J.V., Carter J.A., 2001, Extended Insights into the metastable liquid region behavoir in an adiabatic capillary tube, HVAC&R Research, vol 7, no. 2, p107-123.